UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 7,404,286 B2
APPLICATION NO. : 10/518767
DATED : July 29, 2008
INVENTOR(S) : David Lior

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

Title Page, illustrating a Figure, should be deleted, and replaced showing 19 Drawing Sheets.

Column 6, line 19, add

Fig 11 are OCN-Cross sections of Turbo-shaft version;

Fig 12 is a T-S Diagram;

Fig 13 shows OCN-Thermal efficiencies vs. Compressor pressure ratio;

Fig 14 shows OCN-Specific Power vs. Compressor pressure ratio;

Fig 15 is an OCN and conventional gas turbine Specific power-comparison;

Fig 16 is an OCN and conventional gas turbine efficiencies-comparison;

Fig 17 shows OCN Turbofan-S. F. C. vs. Turbine temperature;

Fig 18 shows OCN Turbofan-Thrust vs. Turbine temperature;

Fig 19 shows an OCN-Effect of Part Load on Thermal Efficiency;

Fig 20 shows an OCN-Effect of Part Load on Power; and

Fig 21 shows OCN-Velocity Triangles.

Signed and Sealed this

Fifteenth Day of September, 2009

David J. Kappos

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Director of the United States Patent and Trademark Office

(12) **United States Patent**
Lior

(10) **Patent No.: US 7,404,286 B2**
(45) **Date of Patent: Jul. 29, 2008**

(54) **ORBITING COMBUSTION NOZZLE ENGINE**

(75) Inventor: **David Lior**, Herzeliya (IL)

(73) Assignee: **R-Jet Engineering Ltd.**, Herzeliya (IL)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 537 days.

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PCT Pub. Date: **Jan. 8, 2004**

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F02C 3/14 (2006.01)
F02C 3/34 (2006.01)
F02C 7/18 (2006.01)

(52) **U.S. Cl.** **60/39.35**; 60/39.34; 60/726; 60/750

(58) **Field of Classification Search** 60/750, 60/806, 39.34, 39.35, 726
See application file for complete search history.

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* cited by examiner

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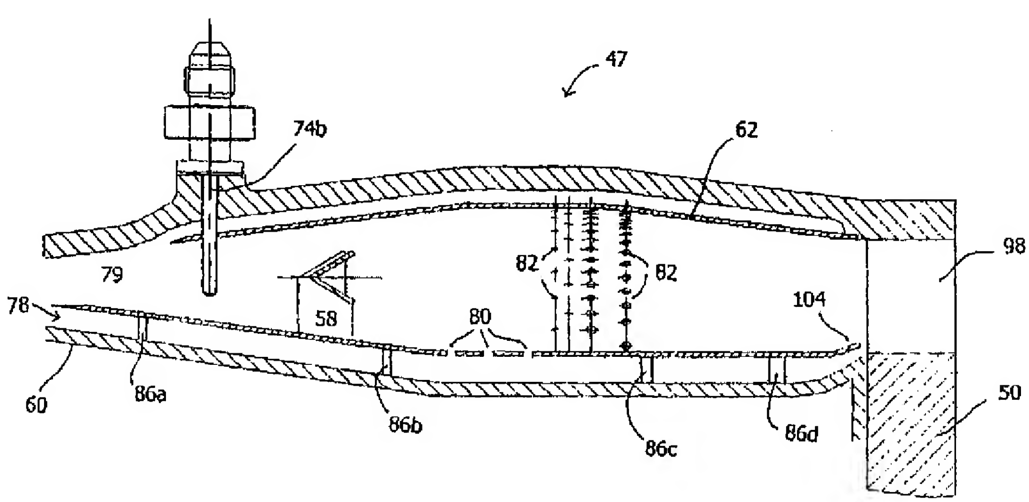
(74) *Attorney, Agent, or Firm*—Mark M. Friedman

(57) **ABSTRACT**

An orbiting combustor nozzle (OCN) engine, having a rotating assembly comprising a co-rotating compressor and nozzle wheel enclosed within a non-rotating outer casing, defining a rotating combustion chamber, is disclosed. Combustion occurs in the combustion chamber in a vortex of gas that rotates at the same angular velocity as the rotating assembly. Also disclosed, is a method of cooling a blade of a rotating wheel, such as a turbine wheel or nozzle wheel, by projecting cool air at the base of the vane form a nozzle corotating with the blade. Such cooling is easily implemented in an OCN engine with use of an innovative annular combustor. Also disclosed is a method of countering axial backflow by use of a combustion chamber compressor.

3 Claims, 19 Drawing sheets



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Column 16, line 38, add

A new concept of an orbiting combustion nozzle (OCN) engine is presented in which the power is provided by a rotating combustion chamber expanding through rotating nozzles, generating a continuous torque and rotating together as one unit. The air is supplied to the combustion chamber from a compressor rotating with the combustion chambers in the same angular velocity, eliminating the conventional stationary compressor diffuser and turbine nozzle guide vanes. A compact engine is thus attained, having the low pollution and continuous combustion advantages of a gas turbine with fewer components and more cost effective.

A thermodynamic analysis results in specific power and thermal efficiencies higher than those of conventional gas turbines while using combustion STATIC temperature lower by 140°K than contemporary gas turbines. The significance of this on emission and reliability is self-evident.

Also, the part load performance of this engine is superior to a conventional cycle gas turbine which is a great advantage in many applications.

II. Nomenclature

C_p - specific heat at constant pressure

$C_{\bar{p}}$ - average value for the progress range

C - absolute velocity

C_v - specific heat at constant volume

E - energy input

F_r - reaction force

k - C_p/C_v

m - mass flow

M - Mach number

P - pressure

P.R - pressure ratio

R - universal gas constant

T - temperature

u - orbital velocity

w - relative velocity

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Suffixes -

a -ambient

c - compressor

e - exit

d - diffuser

is - isentropic

n - nozzle

R - relative

s - static condition

th - thermal

t - total stagnation condition

u - tangential component of velocity

x - actual conditions at nozzle outlet

2 - compressor outlet conditions

3 - nozzle inlet conditions

4 - nozzle outlet conditions

Greek

η - efficiency

ρ - gas density

Δp - combustion chamber pressure loss

1. Introduction

In a conventional gas turbine cycle, air is compressed by a compressor rotor and its dynamic energy at the compressor exit is diffused by a stationary diffuser. This diffusion creates a pressure loss of about 10% of the rotor total pressure, thus decreasing the compressor efficiency and the net work of the gas turbine.

Further, exiting the diffuser the air is introduced into the combustor in which combustion gas is expanded through the turbine to generate power. Since the combustor is stationary, the gas is accelerated again through stationary vanes to match the rotating blade inlet conditions. In doing so, there is an extra loss of total pressure and a decrease of turbine efficiency mainly due to friction losses and aerodynamic vortices in the zone between vanes and blades. Thus, turbine efficiency is impaired - reaching only 85% in small gas turbines.

The combined losses of the turbine and compressor efficiencies result in a reduced performance of the gas turbine - up to 35% reduction (for high pressure ratio cycle) in net power, compared to the OCN performance,

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which due to its unique design eliminates the above losses.

Furthermore, to avoid losses due to shock waves, the conventional turbine is not designed to operate in supersonic inlet blade conditions, thus the pressure ratio per turbine stage is limited (about 2.5). Consequently, in conventional high pressure ratio turbines, there are several stages - stationary and rotating, while the OCN, having no inlet guide vanes and not limited to expand higher pressure ratios with a high efficiency up to 4:1 with one stage, has fewer expansion stages (turbines).

2. Description – Fig. 11

Ambient air is sucked into a compressor [1]. The air is compressed to the desired pressure and rotational speed in axial stages and then in a centrifugal stage.

Air exiting from the compressor rotor [2] is not diffused to stationary conditions but fed through rotating vanes into a rotary combustor [4] thus:

- Eliminating pressure loss as in a conventional diffuser, resulting in higher compressor efficiency, (gain of about 5% for a pressure ratio of 20:1).
- Reaching a higher pressure ratio with the same number of compressor stages compared to a conventional compressor - due to the low relative velocity of the compressor exit flow. Usually in conventional compressors, the pressure ratio is limited to avoid supersonic flow in the diffuser inlet.

Air exiting into the rotary combustor is mixed with fuel and the mixture is burnt in lower static pressure then in a conventional cycle. The swirling air helps in vaporizing the fuel. Its relative velocity is kept low by choosing carefully the compressor outlet conditions. Combustion efficiencies are designed to be between 98-99.8% and pressure drops less than 6% of the inlet relative pressure.

The hot gases are expanded now through rotary nozzles [5] which provide the energy to drive the compressor - a much less enthalpy drop is required than from a conventional turbine due to the higher expansion efficiency in the rotating nozzles. No stationary vanes are needed to expand the hot gas from stagnation conditions into rotating blades. This results in:

- A high adiabatic efficiency of the rotating nozzle - over 90%.
- Due to the high pressure ratio capability of the rotating nozzle - one stage is required for a pressure ratio of 4:1.
- The combustion chamber static temperature is lower than in a conventional cycle (about 125°C lower) for the same power output Fig. 11.

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Exiting the rotating nozzles the gases have a certain swirl which is straightened by stationary diffuser vanes as in a conventional gas turbine.

The exit velocity depends on the cycle parameters and the engine type. In high pressure ratio turbojet and turbofan applications the pressure behind the rotating nozzle is kept at about 2 bars by adding a thrust nozzle, and thus the exit mach number is kept subsonic while in high pressure ratio turbo-shafts exiting into ambient conditions the flow may be transonic. In this case a power turbine [8] may be added and the OCN engine would serve as a gas generator.

3. Thermodynamic Cycle Analysis

Appendix A details the thermodynamic analysis. The equations derived are used to calculate the performance of the engine as detailed in the various performance curves.

Fig. 12 shows the OCN cycle in the T-S diagram, in comparison to conventional cycle. The total pressure of the compressor is kept the same for both cycles [this results in higher total temperature for the conventional cycle due to its lower compressor efficiency.]

Obviously, axial compressor stages are added in front of the centrifugal, the latter is limited to a pressure ratio of 8:1 due to mechanical strength limitations.

Two different OCN cycles are analyzed and compared to the conventional cycle-[A-B-C-D]

- A-B1-C1-E-D1- Heating the gas in the 2 cycles [having the same compressor pressure ratio of 20] to the same total temperature [1300°K] results in higher power output for the OCN cycle. This is obvious from the larger net area in the T-S diagram which results from the higher compressor and turbine efficiencies. Since the heat input for the two cycles is about identical, the net result is higher efficiency [34% against 29%] and higher specific power [210kW against 181 kW] for the OCN cycle.
- A-B2-C2-F-D2- Heating the gas in the OCN cycle to a total relative temperature identical to the stagnation inlet turbine temperature in the conventional cycle, a higher turbine enthalpy drop is obvious in the diagram for the OCN cycle. This is due to its higher efficiency. Even though the heat input is higher, the net work is higher for the OCN cycle which makes it more efficient. [Efficiency is now 35% and specific power is 256 kW]. The detailed analysis further shows it clearly, as may be calculated fromFig. 11for this specific case.

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4. Performance Analysis

Figs. 13, 14 show the design point performance of the OCN cycle for a gas turbine with an airflow of 1 kg/sec and a relative nozzle inlet temperature of 1000°K and 1300°K. Compressor efficiency is 5% higher than in conventional compressor efficiency for the same pressure ratio. Nozzle expansion efficiency is also 5% higher than in conventional turbines for the same expansion ratio. The variable parameters are P.R - total compressor pressure ratio, and u - nozzle orbital velocity.

It is evident from these figures that there is an optimum value of u for a fixed P.R which results in the maximum efficiency and another value for maximum specific power. Taking a design point of u = 500 m/sec (an acceptable value for superalloys), Cu=400m/sec, a nozzle inlet relative temperature of 1300°K and a P.R of 20, the net thermal efficiency is 35% and the power is 256 kW/kg/sec.

Increasing the pressure ratio up to 36 [by adding more compressor stages in front] while the rotating velocity is 600 m/sec results in an efficiency of 38.3%. Figs. 15, 16 depict the OCN versus the conventional cycle performance in various turbine [nozzle] inlet temperatures. For example [see C , D] a conventional cycle with the same P.R and the same turbine inlet stagnation temperature of 1300°K, but with 5% reduced efficiencies for both compressor and turbine the performance is: Power - 181 kW/kg/sec; Efficiency - 29%.,compared to 35% and 256kw of the OCN engine.

Figs. 15, 16 also show the OCN and conventional cycle performance when the speed [and pressure ratio] are constant and the inlet temperature changes. In the OCN cycle the efficiency drops mildly [from 35 to 29%] with the specific power when the temperature drops from 1300°K to 1000°K, while the efficiency of the conventional cycle for the same temperature reduction drops to 8% [see E, F]. This is a significant advantage of the OCN cycle in reduced temperatures in contrast to the poor efficiency of conventional gas turbines.

Analyzing the cycle for high pressure ratio turbojet or turbofan engineFigs. 17, 18 we arrive at the same relative improvement compared to a conventional cycle for specific thrust and S.F.C. values. This superior performance coupled with reduced weight and cost make this engine far more cost effective than a conventional gas turbine.

The advantage of the OCN engine compared to the conventional cycle is thus significant in nozzle inlet temperatures between 1000°K - 1400°K. [Actually, even at inlet temperature of 1600°K the efficiency gain is still 2.5%].

- The thermal efficiency is higher by 4%-21% [absolute value]

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- The specific power is higher throughout the full temperature range by 50-100 %
- The low temperature performance is attractive compared to a conventional cycle.
- The static turbine [nozzle] inlet temperatures are lower by 140°K for the same power requirement. (See Fig. 15 line D-D').

5. Part load performance

Figs. 19, 20 describe the part load performance of an industrial OCN gas turbine with a free turbine when its design point is:

Compressor P.R = 24

Turbine relative total inlet temperature = 1300°K

Airflow = 2.7 kg/sec

Decreasing its turbine inlet temperature the gas generator main shaft speed decreases too, while its free turbine speed is kept constant.

Due to the high adiabatic efficiencies of compression and expansion there is only slight decrease of thermal efficiency [from 35% to 27%] when the load decreases to 30% of its load at 1300°K. In the conventional cycle the thermal efficiency drops to 17% for 30% load.

The above advantages decrease with higher total temperature or in lower pressure ratio. This makes the OCN engine attractive for industrial turbines for electrical energy generation where the long life requirements dictate low turbine inlet temperature, for heavy vehicular use where the part load efficiency is most important and for small efficient aircraft engines where the size and weight are important, a market dominated for over a 100 years by heavy piston engines.

Conclusions

1. The OCN engine cycle is superior to the conventional cycle up to a turbine [nozzle] inlet temperature of 1600k both in specific power and efficiency. This advantage decrease in higher temperature or lower pressure ratio.
2. The OCN engine offers a solution to a new power propulsion concept, presenting a compact configuration , having a specific power and thermal efficiency better than conventional gas turbines.
3. Due to its higher compression and expansion efficiencies the OCN engine thermal efficiency is high even at a 30% load, which is a considerable advantage compared to conventional gas turbines.

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4. The OCN engine delivers the same power as a conventional gas turbine with a lower turbine [nozzle] inlet temperature of about 140°K.
5. Having fewer compressor and turbine stages for the same total pressure ratio the OCN engine has lower weight, less volume, and lower cost.
6. The OCN engine is thus a better cost effective engine suitable to various applications .In particular, due to its flat curve, the OCN engine is a better power plant for small aircraft, gas turbines, and vehicular use such as cars and trucks.

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OCN PERFORMANCE CALCULATION

Power

The power is derived by computing first the reaction force Fr which is:

$F_r = m\ (w_{4u} - w_{3u})$

where:

m - the nozzle exhaust flow rate

u - the rotating tangential nozzle velocity

w_{4u} - the exhaust gas tangential velocity

c_u - absolute tangential nozzle inlet velocity

$w_{3u} = u_3 - C_{3u}$ - relative tangential nozzle inlet velocity.

The turbine power is the product of

$F_r \times u$

$P_n = m\ (w_{4u} - w_{3u})\ u$

and for $w_{3u} = 0$,

$P_n = m\ w_{4u}\ u$ (1)

Note:

In the case of an axial inlet into the nozzle - in the relative space - $W_{3u}=0$, and we carry further the calculation with this assumption, but it can be shown that the calculation result for the value of w_{3u} is identical for any value of w_{3u}.

The net power is derived by subtracting the compressor power P_c from the nozzle power P_n

$P_{net} = P_n - P_c$

The compressor power P_c, for m = 1 kg/sec is derived by calculating the enthalpy change across the compressor:

$$P_c = \frac{C_{\bar{p}}(T_{2t} - T_{1t})}{\eta_m} =$$

$$= \frac{T_{1t}}{\eta_c \eta_m}\left[\left(\frac{P_{2t}}{P_{1t}}\right)^{\frac{K-1}{K}} - 1\right] \qquad (2)$$

in Which:

η_c - adiabatic compressor efficiency

η_m - mechanical compressor efficiency

T_{1t} - compressor inlet total temperature

T_{2t} - compressor outlet total temperature

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P_{1t} - compressor inlet total pressure
P_{2t} - compressor outlet total pressure

The nozzle power P_n is also derived from thermodynamics as follows:-

$$P_n = C_{\bar{p}}(T_{3t} - T_{4t}) \quad (3)$$

in Which;

T_{3t} - nozzle total inlet temperature
T_{4t} - nozzle total outlet temperature.

may also be expressed by:

$$T_{3t} = T_{3R} + \frac{u^2}{2C_{\bar{p}}} \quad (4)$$

$$T_{4t} = T_{4s} + \frac{(w_4 - u)^2}{2C_{\bar{p}}} \quad (5)$$

Where:

T_{4s} - static temperature at nozzle outlet
T_{3R} - relative total temperature at nozzle inlet.

Thus:

$$P_n = \left[\left(T_{3R} + \frac{u^2}{2C_{\bar{p}}}\right) - \left(T_{4S} + \frac{(w_4 - u)^2}{2C_{\bar{p}}}\right)\right] (6)$$

Combining Eqs. (1) and (4) results in:(for 1kg/sec]

$$(w_{4u})u = P_n =$$

$$= C_{\bar{p}} \cdot \left[T_{3R} + \frac{u^2}{2C_{\bar{p}}} - T_{4s} - \frac{w_4^2}{2C_{\bar{p}}} + \frac{(w_{4u})u}{C_{\bar{p}}} - \frac{u^2}{2C_{\bar{p}}}\right] \quad (7)$$

$$= C_{\bar{p}} \left[(T_{3R} - T_s) - \frac{w_4^2}{2C_{\bar{p}}} + \frac{(w_{4u})u}{C_{\bar{p}}}\right]$$

$$= C_{\bar{p}}(T_{3R} - T_{4s}) - \frac{w_4^2}{2} + (w_{4u})u$$

If the exhaust velocity is tangential then

$$w_{4u} = w_T \quad ;$$

and

$$\frac{w_4^2}{2} = C_{\bar{p}}(T_{3R} - T_{4s}) \quad (8)$$

Including the nozzle efficiency results in:

$$w_4^2 = (T_{3R} - T_{4is}) \cdot 2C_{\bar{p}} \cdot \eta_n \quad (9)$$

Where T_{4is} = isentropic static temperature at nozzle exit.

Calculating T_{4is} is done by evaluating the diffuser performance. Total pressure at the diffuser outlet is the

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ambient pressure. Assuming the exit velocity is null and the diffuser efficiency is 100%:

$$p_o = p_{4s}\left(1+\frac{k-1}{2}M_4^2\right)^{\frac{k}{k-1}} \qquad (10)$$

Where M_4 is the local static Mach number at the diffuser inlet and is derived from its definition:

$$M_4 = \frac{(w_4-u)}{\sqrt{kRT_{4s}}} \qquad (11)$$

Equation (9) may be expressed as a function of the pressure ratio across the nozzle

$$\frac{w_4^2}{2C_{\overline{p}}} = T_{3R} - T_{4s} = T_{3R}\cdot\left(1-\frac{T_{4s}}{T_{3R}}\right) =$$

$$= T_{3R}\left[1-\left(\frac{1}{p_{3R}/p_{4s}}\right)^{\frac{k-1}{k}}\right] \qquad (12)$$

Combining Eqs. (10), (11) and (12) results in a single expression for the nozzle exit velocity:

$$w_4^2 = 2C_p\eta_n T_{3R}\left[1-\frac{\left(\frac{p_a}{p_{3R}}\right)^{\frac{k-1}{k}}}{1+\frac{k-1}{2}\frac{(w_4-u)^2}{kR\left(T_{3R}-\frac{w_4^2}{2C_{\overline{p}}}\right)}}\right] \quad (13)$$

Which may be solved mathematically once the values of T_{3R}, u are chosen and introduced.

The diffuser efficiency value decrease the calculated nozzle exit velocity w according to the definition of efficiency:

$$\eta_d = \left(\frac{w_x-u}{w_4-u}\right)^2$$

to its actual value w_x.

Hence:

$$w_x = u + \left(\sqrt{\eta_d}\right)(w_4-u) \qquad (14)$$

Now the value of w_x is used to calculate the net power using Eq. (1).

The value of p_{3R} Which is the total relative pressure at the nozzle inlet is calculated by subtracting

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the combustion pressure drop ΔP from the value of p_{2R} Which is the relative total pressure at the compressor exit and is calculated as follows:

$$\frac{p_{2t}}{p_{2R}} = \left(\frac{T_{2t}}{T_{2R}}\right)^{\frac{k-1}{k}} = \left(\frac{T_{2t}}{T_{2t} - \frac{(2c_{2u}-u)u}{2C_p}}\right)^{\frac{k-1}{k}} \quad (15)$$

Where:

p_{2t} - total pressure at the compressor exit and is determined by the choice of the pressure X

T_{2t} - total temperature at compressor exit.

C_{2u} - tangential component of compressor exit velocity.

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Thermal efficiency calculation

$$\eta_{th} = \frac{\text{net power output}}{\text{thermal power input}}$$

The thermal power input is invested in the fuel injected into the combustion chamber raising the temperature from T_{2R} to T_{3R} Where T_{2R} is derived from Eq. (15):

$$T_{2R} = T_{2i} + \frac{u^2}{2C_{\bar{p}}} - \frac{uc_u}{C_{\bar{p}}}$$

Hence the thermal power input is:

$$E_{th} = \frac{C_{\bar{p}}}{\eta_{com}}(T_{3R} - T_{2R})$$

Where η_{com} is the combustion efficiency.

Introducing the net power output from Eq. (1) the thermal efficiency is:

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$$\eta_{th} = \frac{w_s u - P_c}{\left(\dfrac{C_p}{\eta_{com}(T_{3R} - T_{2R})}\right)} \qquad (16)$$

Calculation Sequence

Input – $C.P.R.$; T_{3R}; η_c; η_m; c_{2u}; u;
$\Delta P_{combustor}$; η_N; η_P

I	Find P_c	Eq. (2).
II	Find P_{2R} and P_{3R}	Eq. (15))
III	Find w_4	Eq. (13).
IV	Find w_s	Eq. (14)
V	Find Net power	Eq. (1).
VI	Find η_{th}	Eq. (16).

Fig. 11 – OCN Turbo shaft Engine - Example

Net Power=630 kW; n_1=54,000 rpm; n_2=45,000 rpm
G=2.7 Kg/sec; Compressor P.R.=16; $T_{combustor}$=1260°K; Thermal Efficiency=35%

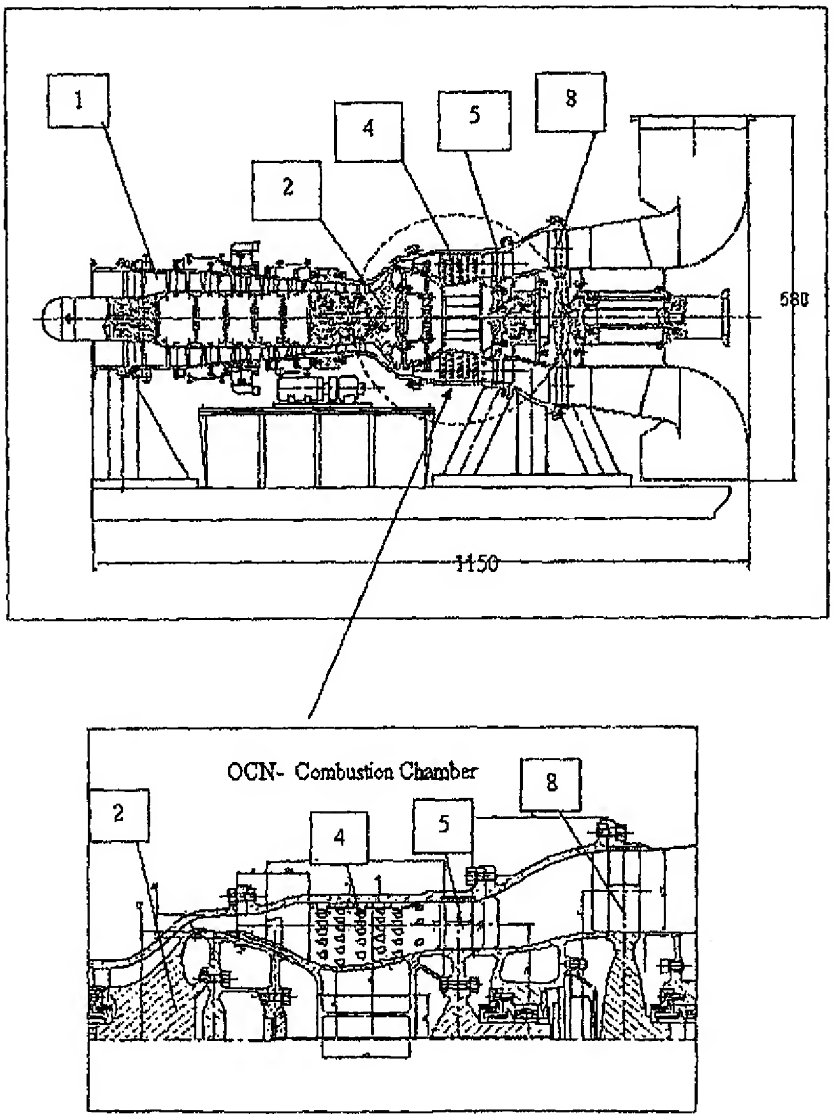
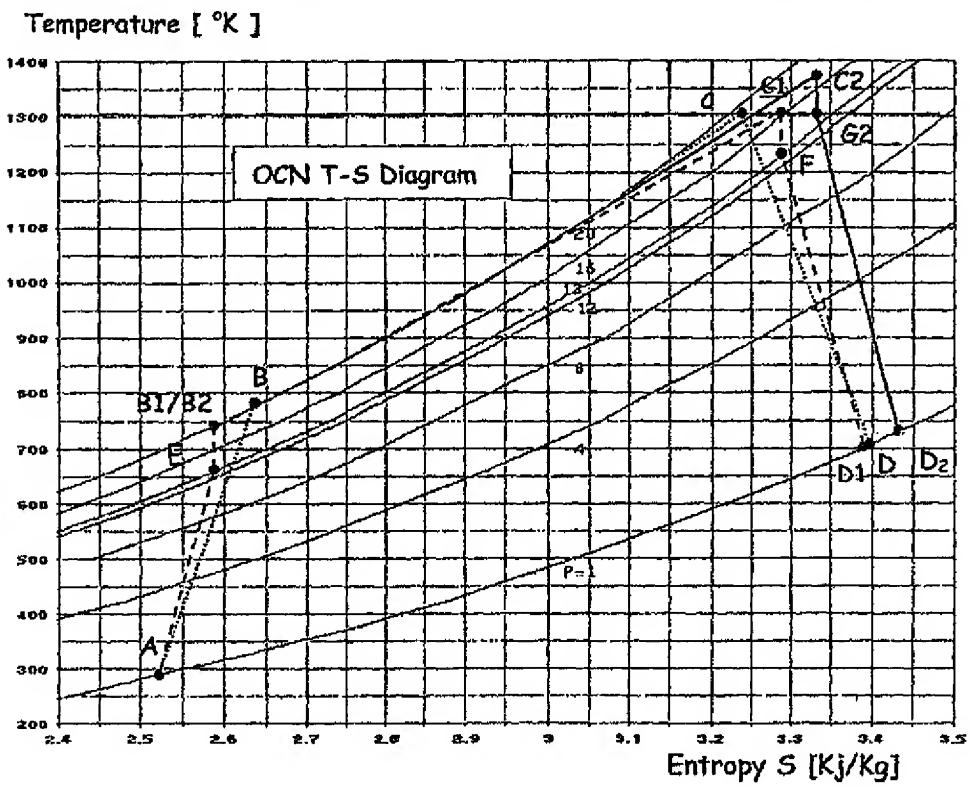


Fig. 12 - OCN T-S Diagram

line style	Cycles	Efficiency	Power	η_c	η_t
....	Conventional = A-B-C-D	29%	181 kW	80%	87%
- - - - - -	OCN, T_1=1300° K = A-E-B1-C1-F-D1	34%	210 kW	85%	92%
————	OCN, T_2=1300° K = A-E-B2-C2-G2-D2	35%	256 kW	85%	92%

Compressor P.R = 20; u = 500 m/sec; Cu = 400 m/sec.



	A	B	B1,B2	C	C1	C_2	D	D1	D2	E	F	G2
T °K	288	777	748	1300	1300	1370	707	702	731	668	1230	1300
P Bar	1	20	20	19.5	15.5	16	1	1	1	13	13	13

Fig. 13 O.C.N THERMAL EFFICIENCIES

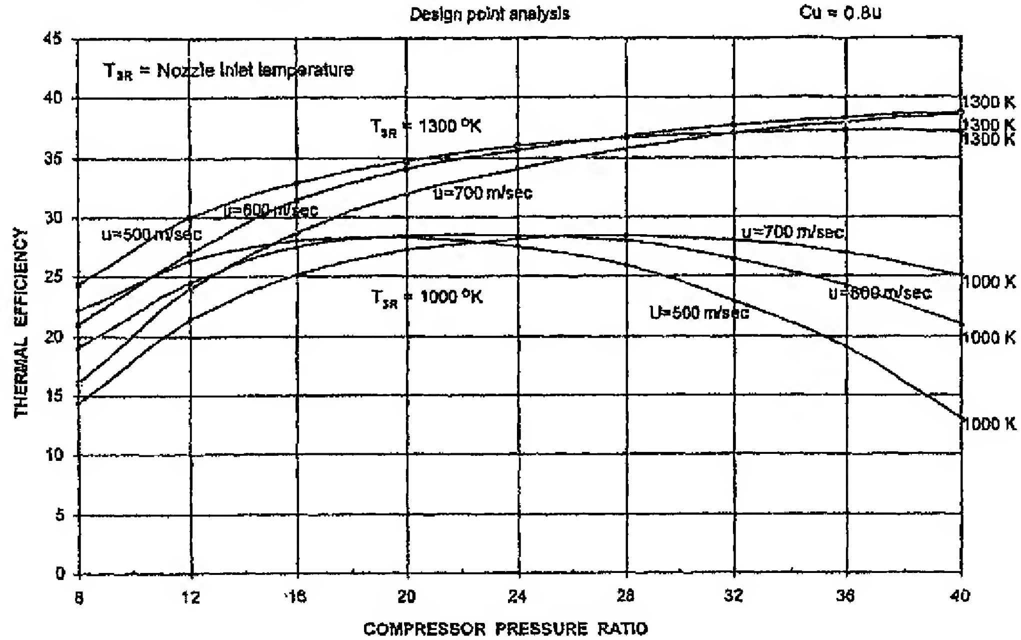


Fig. 14 O.C.N SPECIFIC POWER

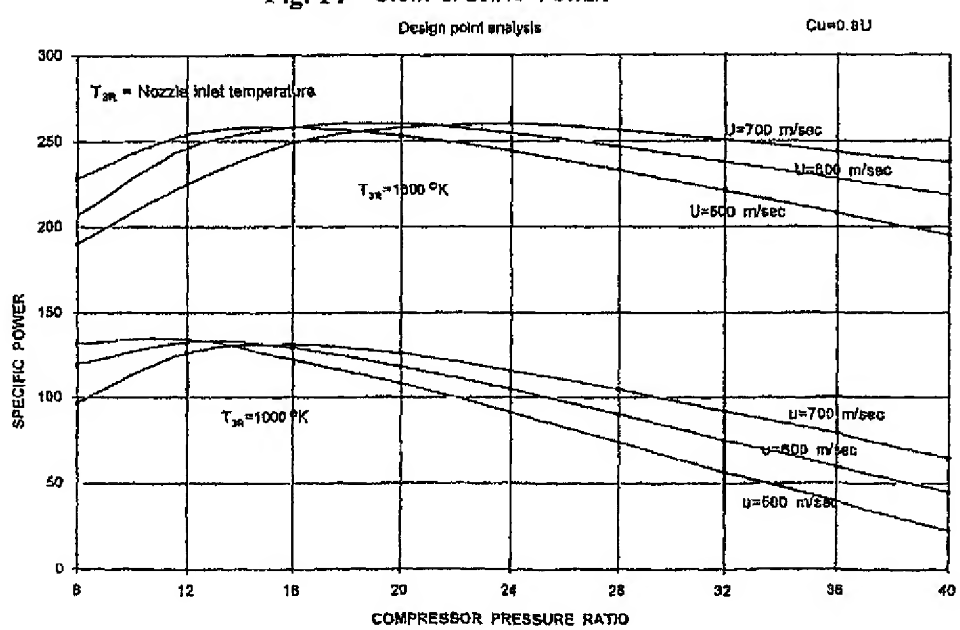


Fig. 15 - O.C.N AND CONVENTIONAL GAS TURBINE SPECIFIC POWER

P.R=20 U=500 m/sec CU =400 m/sec

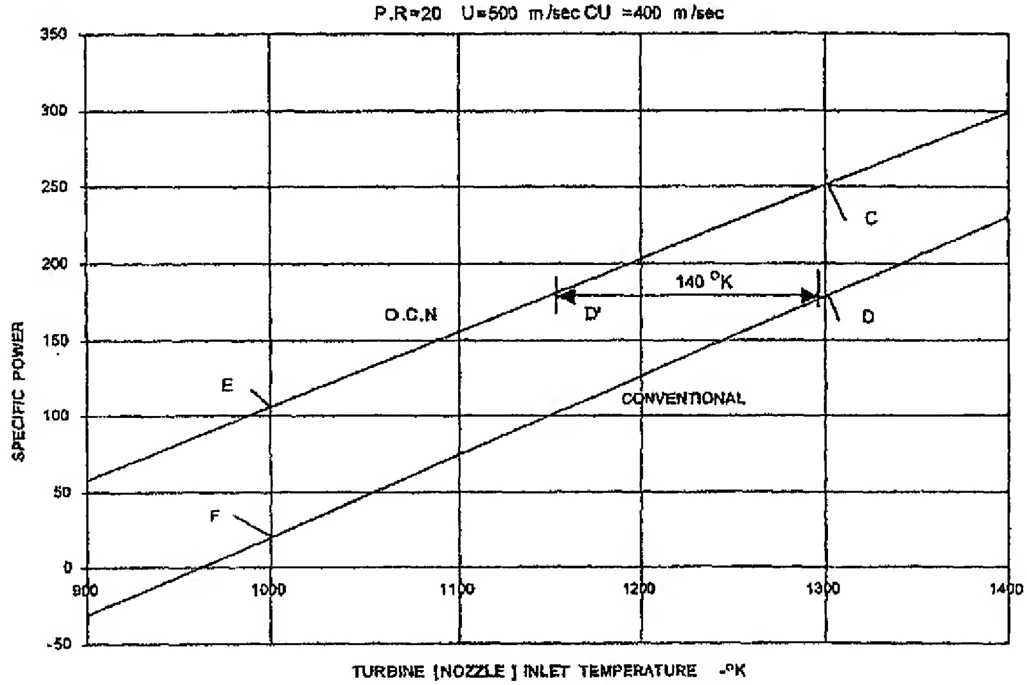


Fig. 16 - O.C.N AND CONVENTIONAL GAS TURBINE EFFICIENC

P.R=20 U=500m/sec Cu=400m/sec

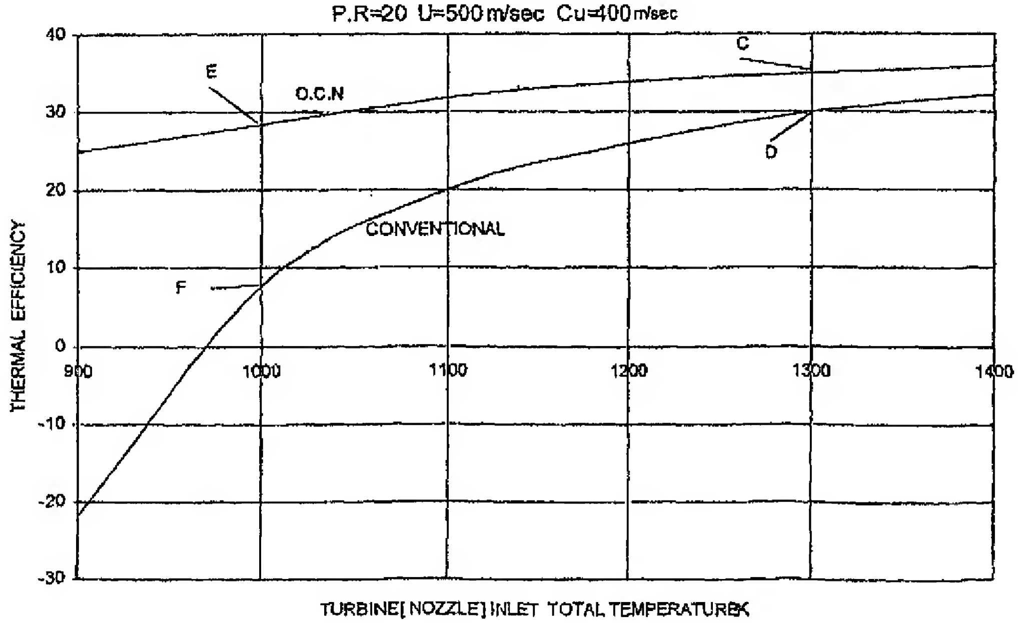


Fig. 17 O.C.N TURBOFAN S .F.C.

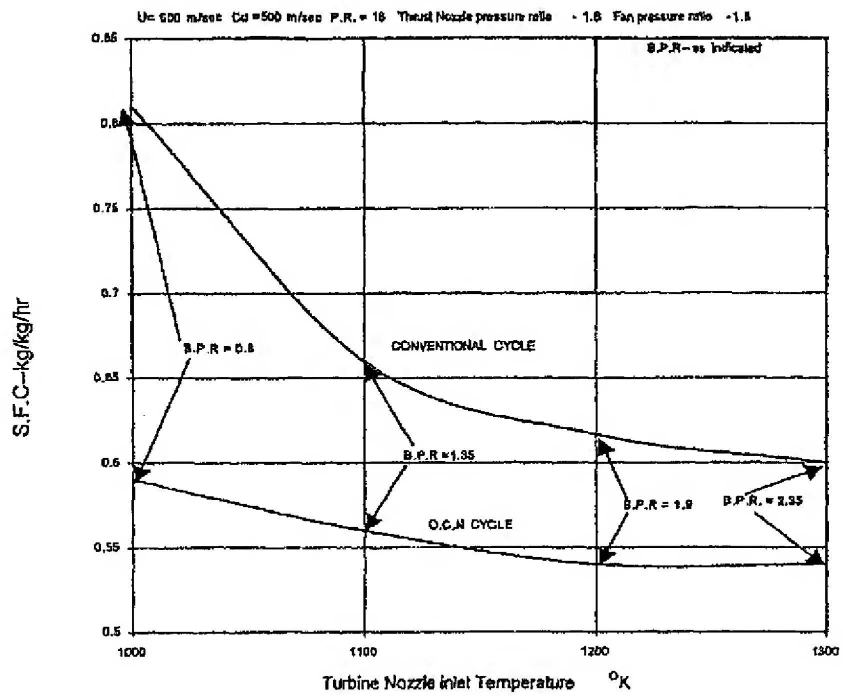


Fig. 18 O.C.N TURBOFAN THRUST

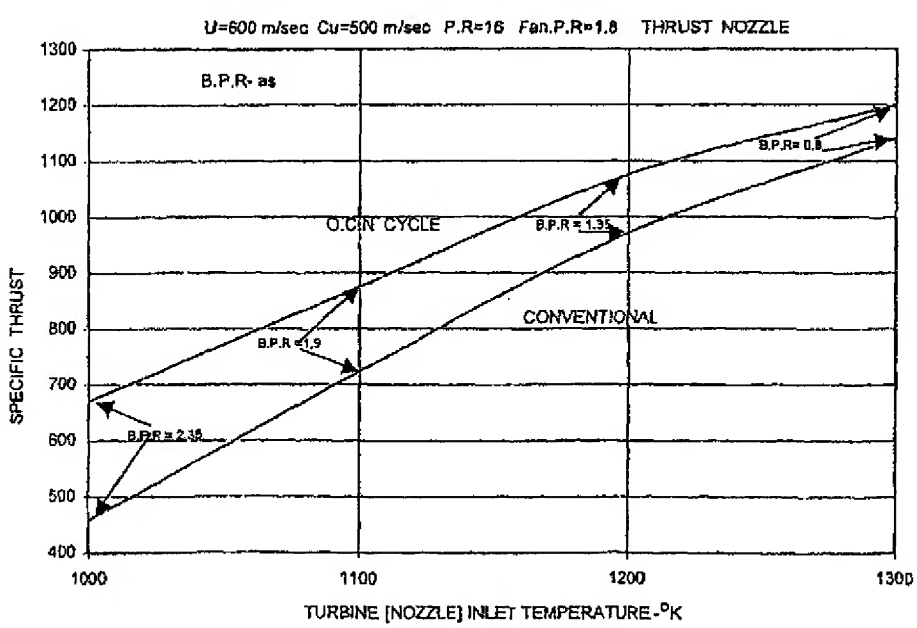


Fig. 19 – EFFECT OF PART LOAD ON THERMAL EFFICIENCY

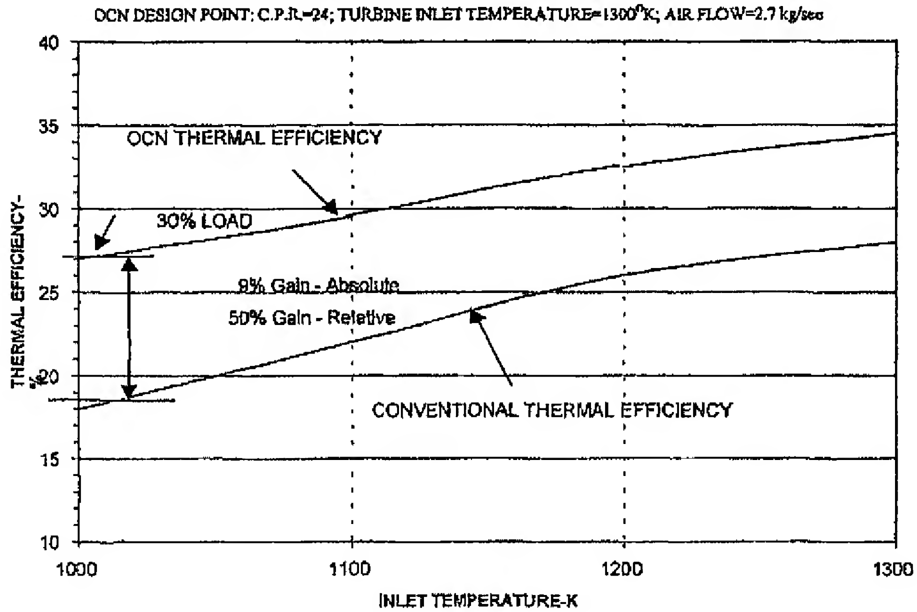


Fig. 20 EFFECT OF PART LOAD ON POWER

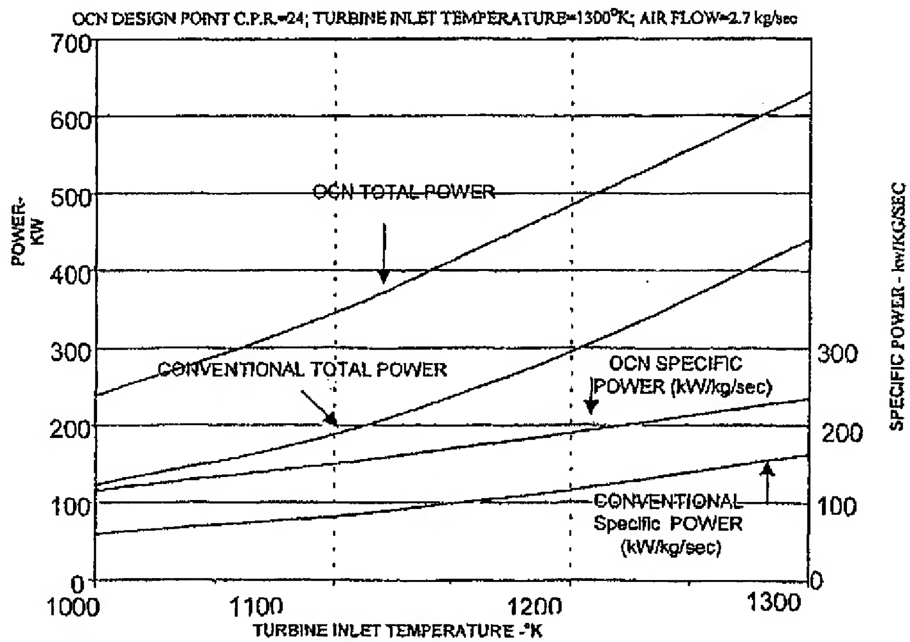


Fig 11. Velocity Triangles

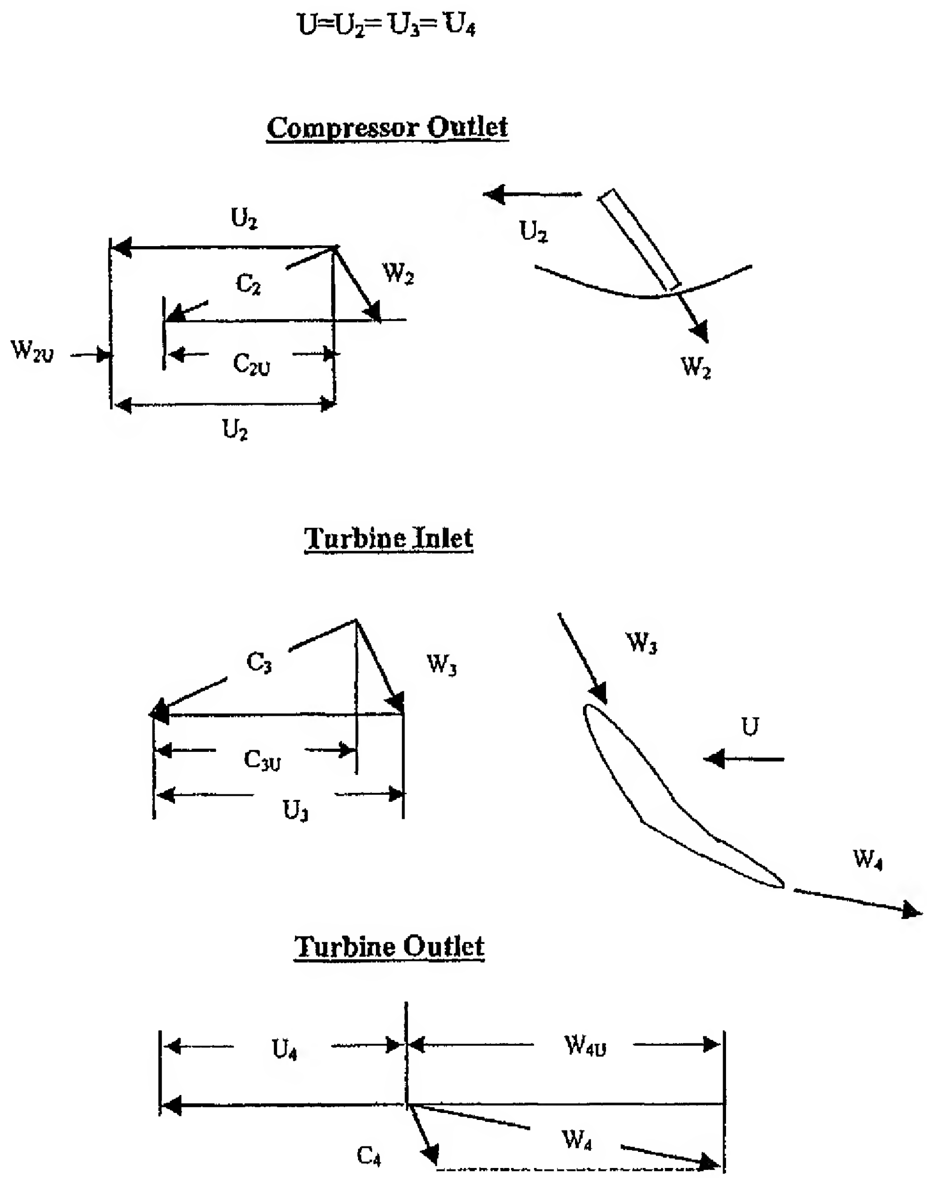

U=U2=U3=U4
Compressor Outlet
U2
C2
W2
W2U
C2U
U2
U2
W2
Turbine Inlet
C3
W3
C3U
U3
W3
U
W4
Turbine Outlet
U4
W4U
C4
W4

Fig. 21